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Investigation of Asymmetrical Vibration Isolators for Maritime Machinery Applications

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ABSTRACT

The development of a novel vibration isolator test facility is described. It is used to measure the dynamic properties of vibration isolators commonly used in maritime and industrial machinery applications. The vibration isolator test facility is capable of measuring the four-pole parameters over the frequency range from 5 Hz to 2 kHz, static load range from 1 to 30 kN, and temperature range from 6 to 60 C.

In order to demonstrate the usefulness of the facility, experimental data for three commercial asymmetrical vibration isolators used for maritime applications are presented. Their four-pole parameters and effectiveness are compared.

RELEASE LIMITATION

Approved for public release

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Executive Summary

The dynamic properties of vibration isolators are dependent upon the parameters of static load, amplitude of vibration, frequency and temperature. It is therefore essential that vibration isolators be tested under service conditions.

This report describes the development of a novel vibration isolator test facility that measures the dynamic properties of vibration isolators commonly used in industrial and maritime applications under service conditions. The vibration isolator test facility is capable of measuring the four-pole parameters over the frequency range from 5 Hz to 2 kHz, static load range from 1 to 30 kN, and temperature range from 6 to 60 °C.

In order to demonstrate the usefulness of the facility, experimental data for three asymmetrical commercial vibration isolators used for maritime applications are presented. The data showed that the measured properties differed from the manufacturer's specification. This may be explained because the testing conditions were not the same for the two sets of data. The reasons for the differences include the effects of different vibration amplitudes, frequencies, temperatures and manufacturing tolerances on the dynamic properties of the rubbers. This demonstrates that vibration isolators must be tested under service conditions, for meaningful measurements.

From the measured four-pole parameters of a vibration isolator, the dynamic properties of any system incorporating it may be investigated. Thus, for example, the effectiveness, force transmissibility and wave effects of a system with a flexible machine and foundation may be determined. The main advantages over other experimental measurements are that the characterisation is independent of the testing method, and the mass effects of the vibration isolator evident at high frequencies are included. These advantages are particularly important for studying the acoustic signatures of naval ships and submarines.

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Contents

Nomenclature

Complex numbers are represented with the superscript *, for example F_1^*, and real numbers are not superscripted.

F_1^*	input force
F_{1R}^*	reversed input force
F_2^*	output force
F_{2R}^*	reversed output force
H_1^*	driving point mobility of source
H_2^*	driving point mobility of foundation
V_1^*	input velocity
V_{1R}^*	reversed input velocity
V_2^*	output velocity
V_{2R}^*	reversed output velocity
f_n	resonant frequency, $n = 1$ to 5
j	$\sqrt{-1}$

Symbols with Greek letters

Ψ^*	effectiveness of isolation
α_{11}^*	four-pole parameter
α_{12}^*	four-pole parameter
α_{21}^*	four-pole parameter
α_{22}^*	four-pole parameter

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1. Introduction

Vibration, including structure borne noise, produced by operating machinery may cause deleterious effects, including excessive stresses in structural or mechanical components, reliability or operational problems with shipboard machines or equipment, and reduced crew efficiency or comfort, AS 3763-1990. For naval ships and submarines, it is important to reduce the vibration transmitted to the hull in order to control the acoustic signature of the vessel.

Effective engineering elements for reducing the vibration and structure borne noise transmitted from shipboard vibratory sources to the supporting structures are vibration isolators. Other terms commonly used for vibration isolators are "anti-vibration mounts", "resilient mounts", "flexible isolators", "vibration mounts" and combinations thereof. A vibration isolator is defined as "a support, usually resilient,"....,"designed to attenuate the transmission of vibration in a frequency range", ISO 2041:1990 and AS 2606-1983.

Resilient elements are commonly rubbers, but also include metal springs, cork, felt and air-bags, Bies and Hansen (1996). Within the rubber industry, the terms "rubber", "polymer" and "elastomer" are used more or less interchangeably for materials having rubber-like properties, ASTM D1566-95a. Rubbers may be natural or synthetic, for example natural rubber is derived from the latex of the botanical source *hevea brasiliensis*, and is synthetically produced as cis-1,4-poly(isoprene).

The effectiveness of a vibration isolator is determined partly by its dynamic properties, yet they are often characterised for industrial applications by their static properties only. Not surprisingly, to be able to design effective vibration isolators, it is necessary to dynamically characterise and test them. The dynamic properties of a vibration isolator depend primarily on the static load of the supported machinery, temperature and exciting frequency and amplitude of vibration. This report describes the development and testing of a novel testing facility for measuring the dynamic properties of large maritime and industrial vibration isolators under service conditions. The facility developed and described in this report is termed the vibration isolator test facility.

Vibration isolators are commonly operated in the compression mode, and this is the configuration primarily tested by the vibration isolator test facility. It is assumed that the orthogonal components of motion for the vibration isolator are uncoupled or only weakly coupled, so that the axial components may be considered independent of the lateral ones. The vibration isolator is considered to comprise a resilient element of homogeneous rubber that is securely bonded to two end plates. The resilient element may be inhomogeneous, for example combinations of different rubbers, and may include intermediate masses. The end plates are used for attaching the vibration isolator to the upper and lower structures. The upper structure is assumed to be the vibratory source such as a machine, and the lower structure the foundation such as the body of a vehicle or the hull of a ship. The upper and lower structures are assumed to present point mobilities to the end plates of the vibration isolator.

Vibration isolators operated in other configurations involving lateral and transverse orientations may also be tested by the vibration isolator test facility. These include inclined orientations employing axial and shear modes of the rubber elements. To conduct the testing two vibration isolators of the same type are required.

An example of the importance of vibration isolators is their application to naval surface ships and submarines, where the control of vibration and structure borne noise transmission is vital for acoustic signature management. High frequencies are important, especially for naval vessels, and of particular concern are the longitudinal standing waves that occur in the rubber elements of vibration isolators. These internal resonances result in the efficiencies of the vibration isolators being significantly reduced with increased transmitted vibration to the hull, and increased radiated noise which is propagated underwater. Jackson et al (1954 and 1956) reported an increase in the transmitted force for a vibration isolator rubber element of loss factor 0.067 of approximately 17 dB with the onset of standing waves. Sykes (1960) reported that for typical natural rubber and neoprene vibration isolators, the effectiveness reduced by as much as 20 or 30 dB at the frequencies of the standing waves. Champagne et al (1997) investigated eight elastomeric materials as possible replacement materials for natural and poly(chloroprene) rubbers used in vibration isolators on Canadian Forces vessels. They modelled the force transmissibilities including standing wave effects, and their results show that at the first standing wave frequency the transmissibility increased by approximately 11 dB for poly(styrene-block-hydrogenated butadiene-block-styrene) which is a thermoplastic triblock copolymer, 11 dB for poly(propylene glycol), 9 dB for natural rubber, 6 dB for poly(epichlorohydrin) and 5 dB for poly(chloroprene) rubber.

Thus the internal longitudinal standing waves of vibration isolators onboard ships and submarines may result in the increased radiation of underwater noise, which may be operationally disadvantageous or intolerable. It is therefore highly desirable to be able to measure the characteristics for the standing waves of vibration isolators under service conditions.

2. Traditional Descriptions

Traditional dynamic descriptions of a vibration isolator have been stiffness, force transmissibility, mobility, impedance and apparent mass. These quantities are usually measured in blocked arrangements, i.e. with the output side of the vibration isolator rigidly supported to give a zero output velocity. For example, Dove and Parry (1973) used stiffness descriptions, Tse et al (1978) and James (1994) employed force transmissibility descriptions, White (1982) applied mobility descriptions, Schloss (1965) used impedance descriptions, and Van Bakel (1986) employed apparent mass descriptions. Damping information was derived from the phase angle, which may be expressed as the loss factor, damping ratio or damping factor. With the phase angles, these measures may be expressed as complex numbers for the stiffness, force transmissibility, mobility, impedance and apparent mass. The mobility (or impedance) of the vibration isolator allows the effectiveness of the vibration isolator to be determined provided that the mobilities (or impedances) of the structures above and beneath the vibration isolator are also known.

3. Characterisation Using Four-Pole Parameters

An alternative dynamic description is provided by the four-pole parameters, which relate the force and velocity at the input of the vibration isolator to the force and velocity at its output. The four-pole parameters are denoted by α_{11}^*, α_{12}^*, α_{21}^* and α_{22}^*. Molloy (1957 and 1958) introduced the four-pole parameter concept to mechanical systems based on four-poles or two-port networks in electrical theory, in which the variables are voltage and current. The concept has also been applied to acoustical systems, with the variables being acoustical pressure and volume velocity. For mechanical vibrations the four-pole parameters are expressed in terms of masses, springs and dashpots, which are analogous to resistances, capacitances and inductances in electrical networks.

There are four main advantages for using the four-pole parameters to dynamically characterise a vibration isolator, as described hereunder.

The first advantage is that the characterisation is independent of the testing method. Consider the traditional description of the transmissibility. It is usually measured by supporting a mass on the vibration isolator to form a single degree-of freedom system, and exciting the mass with a shaker. The vibration isolator is supported by a rigid foundation and its output is considered to be blocked. The transmissibility may then be measured as the ratio of the output force from the vibration isolator, to the input force to the mass. It depends on the supported mass, and is not independent of the test arrangement. This is the reason that the testing machine must be fully described, ISO 2017 and AS 2972.

The second advantage is that mass effects of the vibration isolator evident at high frequencies are included. The traditional descriptions treat a vibration isolator as a massless spring, whereas in reality it has distributed mass and stiffness. The model of a massless spring fails to predict the existence of the longitudinal standing waves found in real vibration isolators. Consider the traditional description of the stiffness, which is usually measured by non-resonant or resonant methods. The non-resonant technique applies a sinusoidal force and measures the resulting displacement, determining the stiffness as the complex ratio of the force to the displacement, ISO 2041 and AS 2606.

These two standards do not clarify if the force is measured at the input or output. At low frequencies the input and output forces are practically equal, but this equality is not true at high frequencies where the mass effects of the vibration isolator are pronounced. If the vibration isolator is assumed to have zero mass, then the input and output forces are equal.

The resonant method determines the magnitude of the stiffness from the resonant frequency of a mass supported on the vibration isolator, and the loss factor from the shape of the resonance curve. This gives the complex value at one frequency. To determine the stiffness at high frequencies requires the use of small supported masses, and the mass of the vibration isolator may become significant in comparison to the supported mass. Under these conditions the mass effects of the vibration isolator will significantly affect the stiffness.

The mass effects for stiffness measurements have not been obvious, because the tests were conducted at low frequencies. For example, the specification MIL-M-17185A (SHIPS) 1959 specifies that the stiffness be measured at the resonant frequencies of the supported masses that apply the upper and lower rated loads to the vibration isolator, and two intermediate masses. By their very nature of operation, vibration isolators in service are designed to have low resonant frequencies, and so the measuring frequencies are low. The dynamic stiffnesses specified by the specifications MIL-M-19379B (SHIPS) 1961, MIL-M-17185A (SHIPS) 1959, MIL-M-17191D (SHIPS) 1970, MIL-M-21649C (SH) 1991 and MIL-M-19863D (SH) 1991 may be measured by resonant or non-resonant methods, but in all cases the measuring frequencies do not exceed 15 Hz.

The traditional descriptions of the mobility, impedance and apparent mass are directly related to the stiffness of the vibration isolator and the frequency, and are usually determined in similar ways as the stiffness. They therefore suffer from the same problem of not including mass effects at high frequencies.

The third advantage is that the four-pole parameters of a mechanical system may be derived mathematically from the four-pole parameters of its constituent series and parallel parts. Thus a complicated system comprising a number of masses, springs and dashpots may be analysed from the characterisations of a mass, spring and dashpot in terms of the four-pole parameters. A simple example is a vibration isolator, that may be modelled as a spring and dashpot terminated at each end by a mass. A more complicated example is a naval system comprising a machine mounted on a raft via vibration isolators, and the raft in turn supported by the hull with vibration isolators. This allows a system to be analysed by breaking it into simpler components, and is analogous to the synthesis of electrical filters and attenuators. This also permits the effect of the output force and velocity of a vibration isolator on a non-rigid supporting structure to be modelled.

The fourth advantage is that the four-pole parameters are amenable to modelling the internal wave effects and the transmission of power. The transmitted power is the product of the force and the complex conjugate of the velocity, and the four-pole parameters are in terms of the forces and velocities.

The investigation of acoustic signatures for naval ships and submarines requires that vibration isolator characterisation be independent of the testing method and include mass effects at high frequencies. Consequently tradition descriptions of vibration isolators are inadequate, and the four-pole parameters description is used in this study.

4. Relationships of Four-Pole Parameters

The rubber in a vibration isolator, being a highly non-linear material, causes it to have non-linear behaviour. In common practice the vibration isolator supports a machine and is subjected to a dynamic force superposed on a static load. Under normal operation, the input dynamic vibration amplitudes are generally much smaller than the compressed height of the rubber. The vibration isolator thus operates dynamically about a static point on its force displacement curve. For small dynamic strains of not greater than $1\text{x}10^{-3}$, the complex normal modulus may be treated as constant, Payne (1956) and Payne and Scott (1960). Under these conditions the dynamic characteristics of a vibration isolator may be considered to be linear with respect to displacement, but still frequency dependent. Therefore a vibration isolator may be dynamically represented as a pseudo-linear system, Figure 1, where the dynamic force and velocity at its input are denoted by F_1^* and V_1^* respectively, and the dynamic force and velocity at its output by F_2^* and V_2^* respectively.

Let α_{11}^*, α_{12}^*, α_{21}^* and α_{22}^* denote the four-pole parameters, which are complex, time invariant functions of the circular frequency ω. The four-pole parameters relate the input and output forces of the vibration isolator and are defined by

$$\begin{bmatrix} F_1^* \\ V_1^* \end{bmatrix} = \begin{bmatrix} \alpha_{11}^* & \alpha_{12}^* \\ \alpha_{21}^* & \alpha_{22}^* \end{bmatrix} \begin{bmatrix} F_2^* \\ V_2^* \end{bmatrix} \qquad (1)$$

Assuming that the Rayleigh's reciprocity theorem in the form of Maxwell's reciprocal deflections theorem applies to the system, it may be shown that, Molloy (1957),

$$\alpha_{11}^*\alpha_{22}^* - \alpha_{12}^*\alpha_{21}^* = 1 \qquad (2)$$

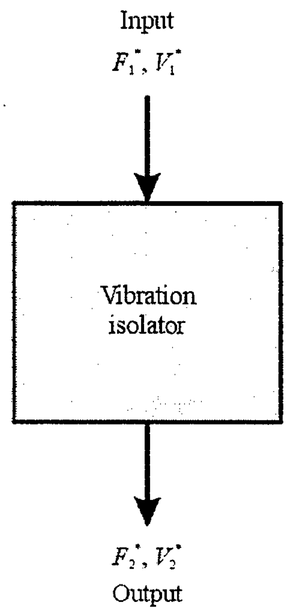

Figure 1 Block representation of a vibration isolator.

4.1 Symmetrical Vibration Isolators

Symmetrical vibration isolators are defined as those that behave the same if the input and output ends are interchanged. For these vibration isolators it may be shown that, Snowdon (1971),

$$\alpha_{11}^* = \alpha_{22}^* \tag{3}$$

4.2 Asymmetrical Vibration Isolators

Asymmetrical vibration isolators are those vibration isolators that do not behave the same if the input and output ends are interchanged. For asymmetrical vibration isolators, equation (3) is no longer valid and so additional information must be obtained. This is normally been done by reversing the vibration isolator in the test rig so that its input and output sides are interchanged, Snowdon (1979a and 1979b). Consider this reversed configuration and denote the input force and velocity by F_{1R}^* and V_{1R}^* respectively, and the output force and velocity F_{2R}^* and V_{2R}^* respectively. Equation (1) then becomes

$$\begin{bmatrix} F_{1R}^* \\ V_{1R}^* \end{bmatrix} = \begin{bmatrix} \alpha_{22}^* & \alpha_{12}^* \\ \alpha_{21}^* & \alpha_{11}^* \end{bmatrix} \begin{bmatrix} F_{2R}^* \\ V_{2R}^* \end{bmatrix} \tag{4}$$

For the blocked situation, $V_{2R}^* = 0$ and so from equation (4),

$$\alpha_{22}^* = \left. \frac{F_{1R}^*}{F_{2R}^*} \right|_{V_{2R}^*=0} \tag{5}$$

and

$$\alpha_{21}^* = \left. \frac{V_{1R}^*}{F_{2R}^*} \right|_{V_{2R}^*=0} \tag{6}$$

Equation (5) provides the additional relationship to determine α_{22}^*, and equation (6) may be used to experimentally check the value of α_{21}^*. For the unblocked situation, equations (1), (2) and (4) may be combined to give, Meltzer and Melzig-Thiel (1980),

$$\alpha_{12}{}^* = \frac{F_1{}^* F_{1R}{}^* - F_2{}^* F_{2R}{}^*}{V_1{}^* F_{2R}{}^* + V_{2R}{}^* F_1{}^*}, \tag{7}$$

$$\alpha_{11}{}^* = \frac{F_1{}^* - \alpha_{12}{}^* V_2{}^*}{F_2{}^*}, \tag{8}$$

$$\alpha_{22}{}^* = \frac{F_2{}^* + \alpha_{12}{}^* V_1{}^*}{F_1{}^*} \tag{9}$$

and

$$\alpha_{21}{}^* = \frac{V_1{}^* - \alpha_{22}{}^* V_2{}^*}{F_2{}^*} \tag{10}$$

This approach of reversing the vibration isolator in the test rig assumes that the vibration isolator is bi-directional and it may be operated with its input and output interchanged.

4.3 Effectiveness

Jacobsen and Ohlrich (1986) showed that the effectiveness Ψ^* of a vibration isolator may be expressed in terms of its four-pole parameters and the source mobility $H_1{}^*$ and foundation mobility $H_2{}^*$, as

$$\Psi^* = \frac{\alpha_{11}{}^* H_1{}^* + \alpha_{12}{}^* H_1{}^* H_2{}^* + \alpha_{21}{}^* + \alpha_{22}{}^* H_2{}^*}{H_1{}^* + H_2{}^*} \tag{11}$$

If the vibration isolator supports a mass *m*, and the foundation has infinite stiffness i.e. zero mobility, then the source and foundation mobilities are given by

$$H_1{}^* = \frac{1}{j\omega m} \tag{12}$$

and

$$H_2{}^* = 0 \tag{13}$$

Substituting equations (12) and (13) into equation (11) gives

$$\Psi^* = \alpha_{11}{}^* + j\omega\alpha_{21}{}^* \tag{14}$$

5. Vibration Isolator Test Facility

A novel vibration isolator test facility that is capable of characterising small and large vibration isolators under different static loads, temperatures and frequencies was developed. It is capable of testing vibration isolators over the frequency range from 5 Hz to 2 kHz, static load range from 1 to 30 kN, and temperature range from 6 to 60 °C. It can test practical vibration isolators having dynamic stiffness magnitudes of at least 1 x 10^5 N/m, masses of at least 0.5 kg, and loss factors of at least 0.03. Thus it is able to investigate vibration isolators commonly used in maritime and industrial applications. The results are presented in terms of the four-pole parameters, as narrow band amplitude and phase data.

Photographs of the vibration isolator test facility are presented in Figures 2 to 6; the first one shows the general layout. The vibration isolator test facility comprises the test rig, instrumentation, shaker and associated power amplifier, and temperature conditioning unit.

5.1 Test Rig

A developmental test rig was used to study experimental methods of measuring the four-pole parameters of vibration isolators, Dickens (1993), Dickens and Norwood (1994, 1995a, 1995b, 1996a, 1996b and 1997) and Norwood and Dickens (1998). The test rig developed by Farquharson (1991) was based on the measurement method of Verheij (1980 and 1982), and was modified and used as the developmental test rig, Dickens (1998).

Consequently the present test rig was developed that implements a novel measurement procedure using knowledge gained from the developmental test rig. The novel procedure incorporates the floating mass method, measures the direct forces and corrects analytically for errors introduced using the direct force method, Dickens (1998).

The test rig comprises two support frames, masses, isolation hangers, table extension, connecting rod, base plate and air-bags, Figures 3 and 4. The test rig is shown schematically in Figure 7. The test rig is primarily designed to test the axial i.e. vertical direction, which is normally the main direction of interest. With appropriate attachments, it can also be used to test the lateral directions. The vibration isolator under test is mounted between two large masses, and static load is applied by air-bags positioned above and beneath the two masses. The dynamic load is applied by an electro-dynamic shaker via the upper mass, and the lower mass provides a reaction force to the output force of the vibration isolator.

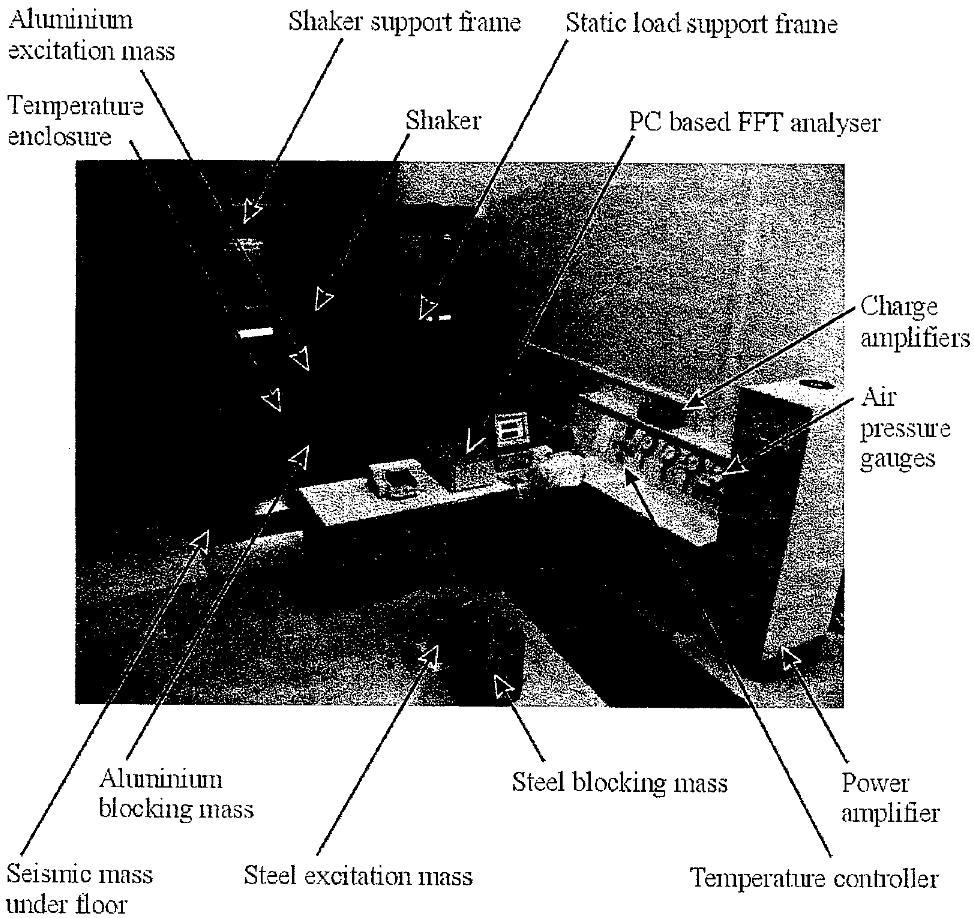

Figure 2 Layout of vibration isolator test facility.

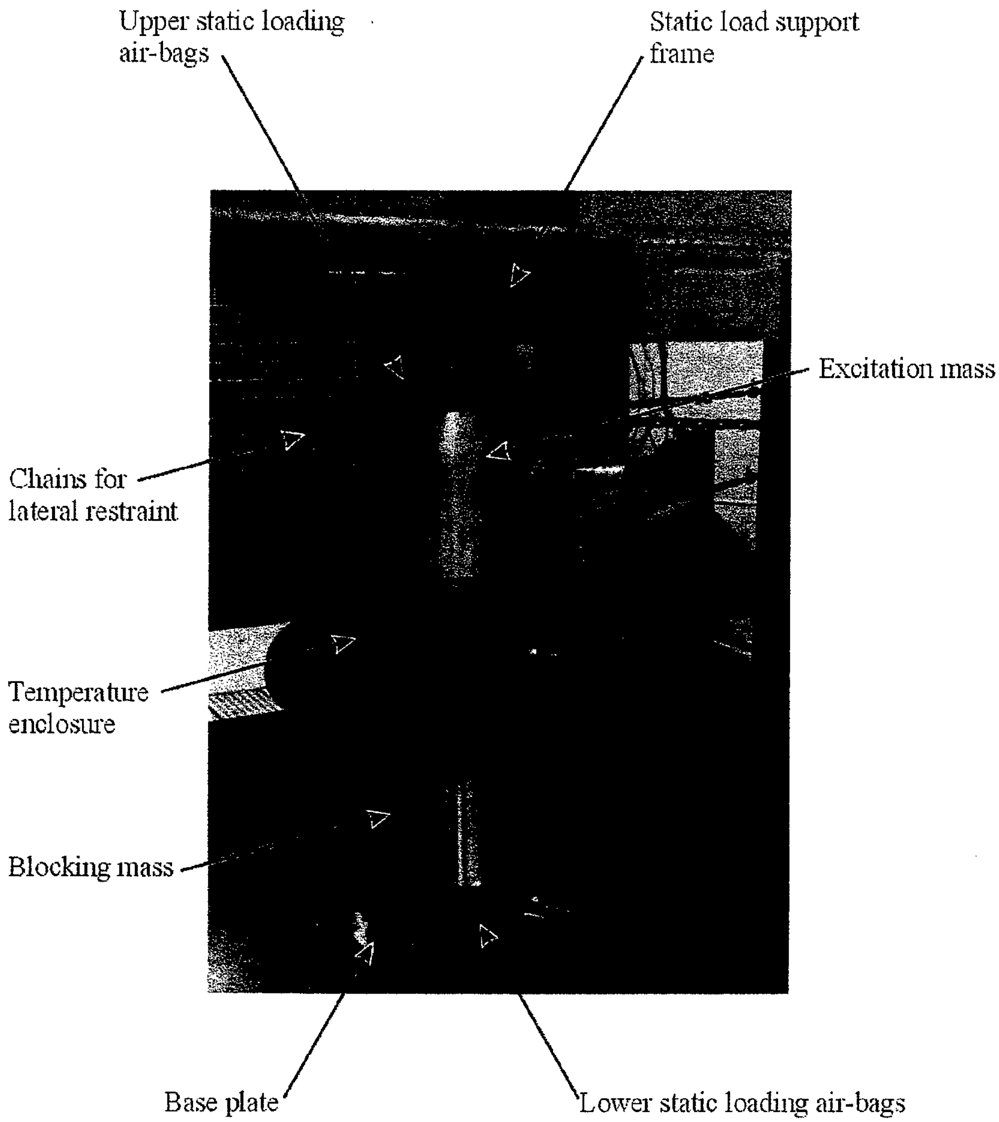

Figure 3 Excitation and blocking masses.

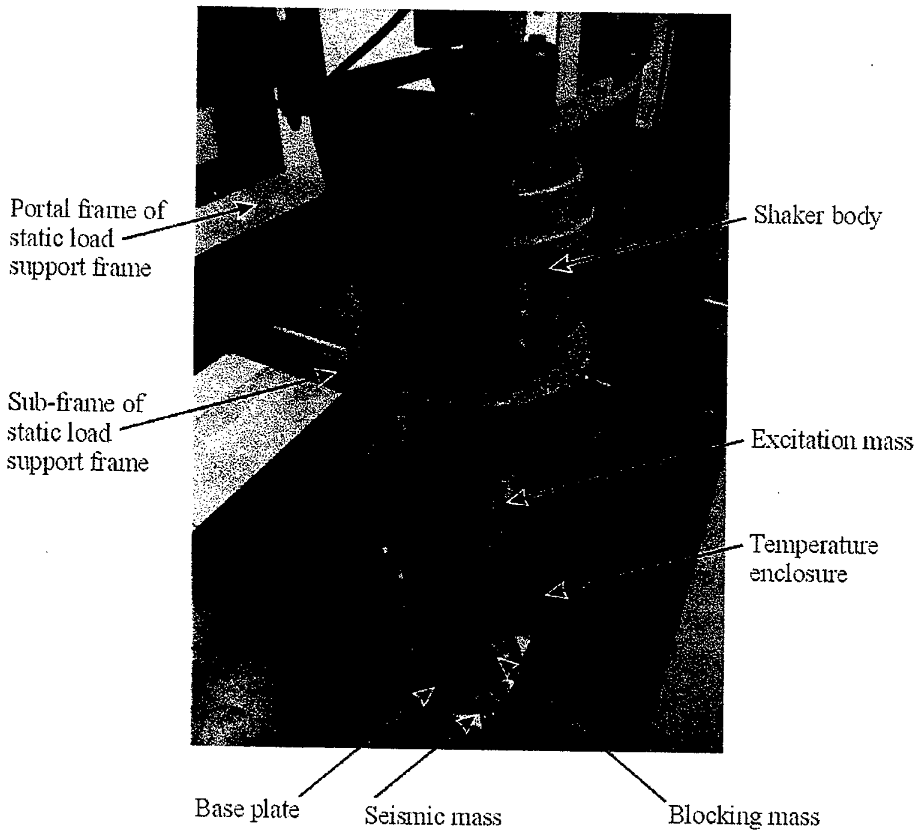

Figure 4 Test arrangement viewed from above.

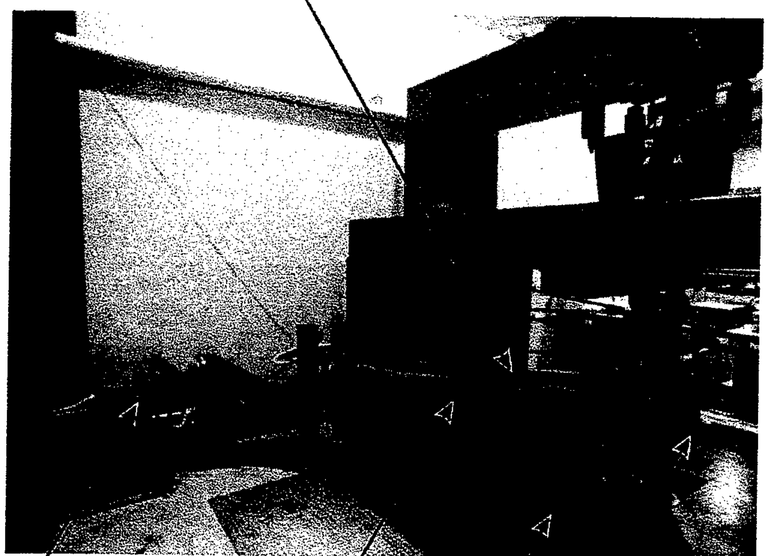

Figure 5 Temperature conditioning unit.

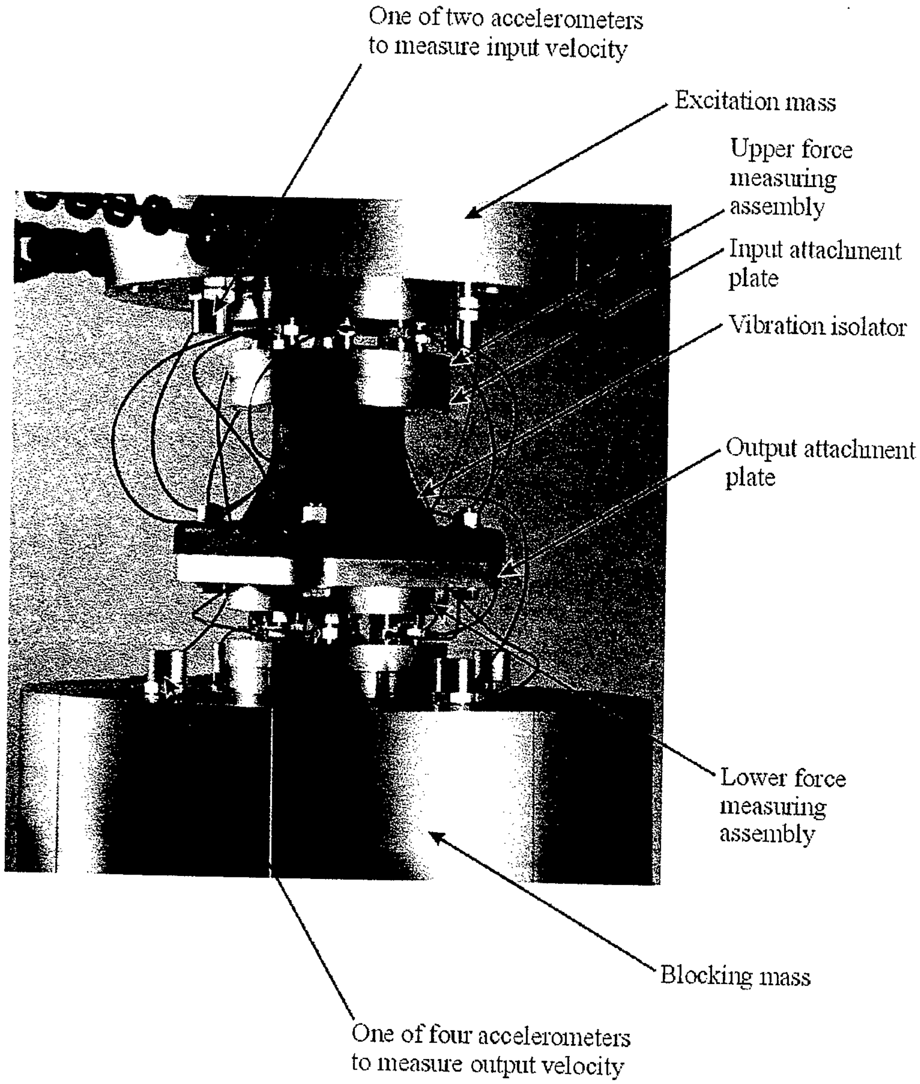

Figure 6 Vibration isolator under test with temperature enclosure removed for clarity.

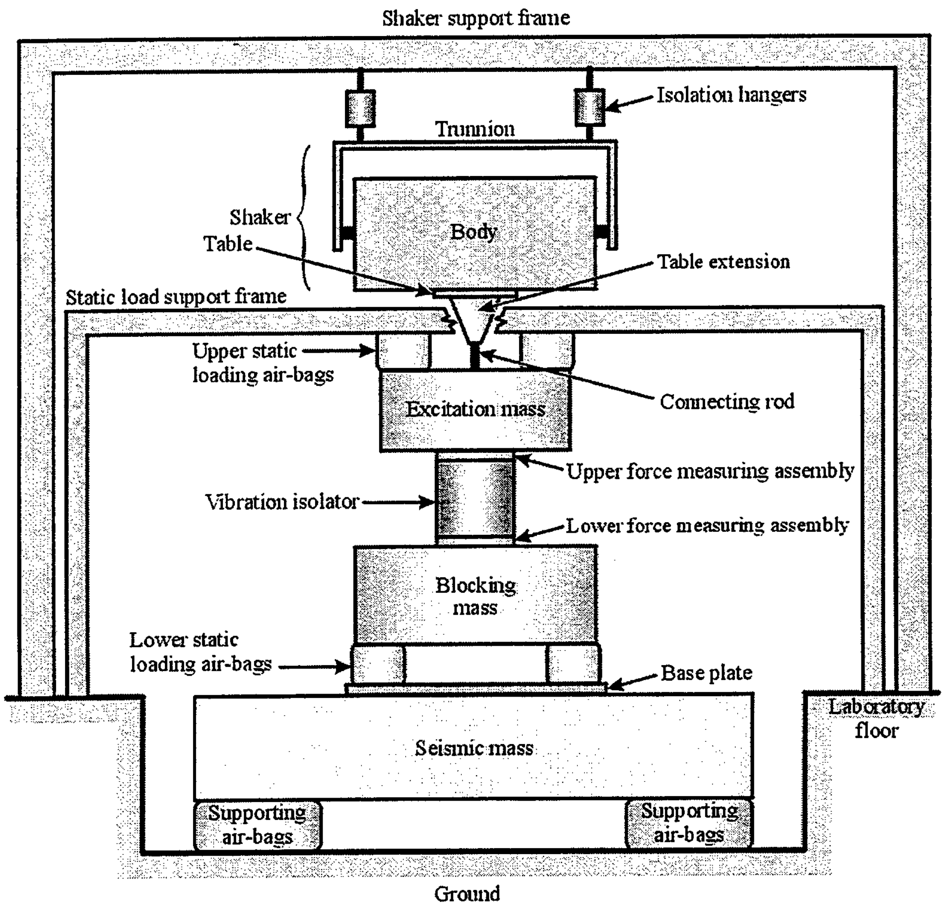

Figure 7 Schematic diagram of vibration isolator test rig.

The upper and lower masses are termed the excitation and blocking masses, respectively. The rig has two supporting frames, the upper frame that supports the shaker and the lower frame that provides the reaction forces for the upper static loading air-bags. The excitation mass is laterally constrained by chains attached to the lower frame. The upper and lower frames are respectively termed the shaker support frame and the static load support frame. The lower static loading air-bags sit on a base plate mounted on top of a seismic mass.

Two frames are used to reduce coupling between the static loading structure and the shaker. The shaker is decoupled from its supporting frame by four isolation hangers and drives the excitation mass through a single centrally located connecting rod. The seismic mass is a block of reinforced concrete of dimensions 3.0 m x 3.0 m x 1.0 m, supported on four air-bags connected to air reservoirs and having a mounted natural frequency of approximately 1.8 Hz. The use of the seismic mass decouples the blocking mass from the laboratory floor, reducing the input of extraneous forces and transmissions from the two supporting frames and the environment.

Air-bags are used to provide the static load as the static force can be easily adjusted, while at the same time giving a degree of isolation between the masses and the supporting structures. The motions of the excitation and blocking masses are measured using accelerometers.

The input force to the vibration isolator is the dynamic force applied by the excitation mass to the vibration isolator and is measured directly by a force measuring assembly. The output force is the dynamic force applied by the vibration isolator to the blocking mass and is determined directly by a force measuring assembly. Each force measuring assembly consists of a parallel arrangement of eight force transducers.

In designing the test rig it was important that the dynamic response of the test rig did not affect the measurement of the vibration isolator's four-pole parameters. This implies that where possible the components of the test rig should not have structural modes within or near the frequency band of interest, and where this is not possible the structural response of the rig should not affect the results. This requirement was met and verified by testing the developed test rig, Dickens (1988).

5.2 Instrumentation

The instrumentation comprises an FFT analyser to control the test and acquire the data, accelerometers to measure the accelerations, and force measuring assemblies to measure the forces. Accelerometers, force measuring assemblies and a vibration isolator under test are visible in Figure 6.

A digital computerised multi-channel Hewlett Packard FFT spectrum/network analyser type 3566A was used to control the test, and to acquire and initially analyse the data. It generated an output signal that controlled the power amplifier supplying the shaker, and so controlled the vibratory forcing function applied to the excitation mass. This control was either open loop, or closed loop by feeding back the monitored excitation mass acceleration to the analyser.

The analyser was programmed to conduct swept sine tests across a selectable frequency band in a selectable number of steps that were logarithmically or linearly spaced.

5.2.1 Measurement of Forces

Two force measuring assemblies were developed to measure the direct input and output forces.

Each developed force measuring assembly consisted of eight Bruel & Kjaer force transducers type 8200 attached between two ground aluminium plates. This gave a total static plus dynamic force capability of 40 kN. In each force measuring assembly, all of the force transducers were matched and had the same charge sensitivity to three significant figures. Therefore their sensitivities were equal to within ±0.13 %. The force transducers were piezo-electric devices having charge outputs. Thus the total force measured by them could be obtained by connecting their outputs in parallel to give a total charge equal to the addition of their individual charges. This was accomplished using a PCB summing unit type 70A15. This meant that their combined charge output could be amplified using only one charge amplifier to give the total force, in amplitude and phase. Additionally only one channel of data acquisition per force measuring assembly was required, instead of eight. Therefore the cost, complexity and data acquisition time of the measurement was significantly diminished.

The static force applied at the interface between the bottom plate of the upper force measuring assembly and the top plate of the vibration isolator was determined by using the output from the upper force measuring assembly. This output was fed into a low frequency Bruel & Kjaer charge amplifier type 2628 set on the longest time constant and lowest upper frequency limit of 100 Hz. The output of the charge amplifier was monitored with a digital voltmeter and the quasi-static force measured. The value of the time constant was very long, and was specified as 1×10^4 s, i.e. 2.78 hours. In practice, the quasi-static force could be measured with no loss of accuracy caused by the decay of the output voltage.

The measured forces were corrected to account for the modal behaviour of the force measuring assemblies at high frequencies.

5.2.2 Measurement of Accelerations

The acceleration of the excitation mass was measured by averaging the signals from a pair of symmetrically positioned Bruel & Kjaer accelerometers type 4379 screwed to the bottom surface of the excitation mass. The accelerometers were piezo-electric devices and were matched, and consequently their charge outputs were added by using a PCB summing unit type 70A15. The accelerometers had the same charge sensitivity to three significant figures and were equal to within ± 0.16%. The combined charge output was fed into a Bruel & Kjaer charge amplifier type 2635. In a similar way the acceleration of the blocking mass was measured from four matched Bruel & Kjaer accelerometers type 4379 screwed to the top surface of the blocking mass with a Bruel & Kjaer charge amplifier type 2635.

The measured accelerations were corrected to account for the modal behaviour of the force measuring assemblies at high frequencies.

5.3 Shaker and Power Amplifier

The shaker and power amplifier are visible in Figures 2 and 4. An MB Dynamics shaker type C10E, and an MB Dynamics Power Amplifier type M24K are used. The shaker has a sinusoidal force rating of 5.3 kN (peak), a stroke of 25.4 mm (peak-to-peak) and an operating frequency range from 5 Hz to 3 kHz. It uses forced air cooling and is capable of inverted operation to suit the set-up of the test rig. It has both internal and external degaussing coils and the operating current of the latter one was adjusted to minimise the magnetic field and consequently its interference with the operation of the transducers.

5.4 Temperature Conditioning Unit

The temperature conditioning unit was developed to condition the test vibration isolator at a constant temperature. It comprises the temperature enclosure and associated support arm, heating/cooling unit, temperature controller, temperature sensors and a temperature recorder. The major components are visible in Figure 5. The vibration isolator is enclosed within an enclosure maintained at a constant temperature over the range from 6 to 60 °C with an accuracy of ±1 °C.

The sides of the enclosure are insulated with 50 mm of thermal insulating fibre material and the top and bottom are lightweight 50 mm thermal insulating material attached to the excitation and blocking masses using Velcro fasteners. There is a small gap between the sides and the top and bottom of the enclosure, which is required during testing to prevent the enclosure from affecting the results. During the temperature conditioning prior to testing, the gap is filled with insulating material.

The enclosure is maintained at a constant temperature with a closed system of heated or cooled recirculating air, which is supplied by a heating/cooling unit with an associated external compressor unit and air hoses. The arm supporting the enclosure is pivoted on a vertical support bolted to the laboratory floor. The enclosure is in two attachable parts, and when required may be rotated away from the vibration isolator on the supporting arm.

The temperature inside the enclosure is monitored using platinum resistance temperature detectors with specified accuracies of ±0.03 % at 0 °C. The temperature is controlled using an RKC Instrument controller type REX-F900 with a specified accuracy of ±0.01 %. The temperatures inside the enclosure are recorded using a Chino six channel temperature recorder type 12AL660NNN.

5.5 Adjustment for Height of Vibration Isolator

To be able to test vibration isolators having different heights, a number of spacers were developed to be bolted between the force measuring assemblies and the excitation and blocking masses. These spacers were steel discs of diameter 170 mm and varying heights. Other spacers were developed for insertion between the lower static loading air-bags and the base plate, and have dimensions of 85 mm diameter x 100 mm height. Modal analyses showed that the modal behaviour of the spacers would not affect the measurements.

In addition to using the spacers, the heights of the air-bags may be varied. The upper static loading air-bags have a specified operating design range of ±25 mm, the lower static loading air-bags have a specified operating design range of ±19 mm, and the air-bags supporting the seismic mass have a specified operating design range of ±50 mm.

Using the spacers and operating ranges of the air-bags, the heights of vibration isolators that may be tested is from zero to 314 mm.

6. Tests of Maritime Vibration Isolators

Three commercial vibration isolators used for maritime purposes were tested with the vibration isolator test facility. The type of vibration isolator tested are used for protecting equipment onboard naval ships, including vessels of the Royal Australian Navy. One of these is shown in Figure 8.

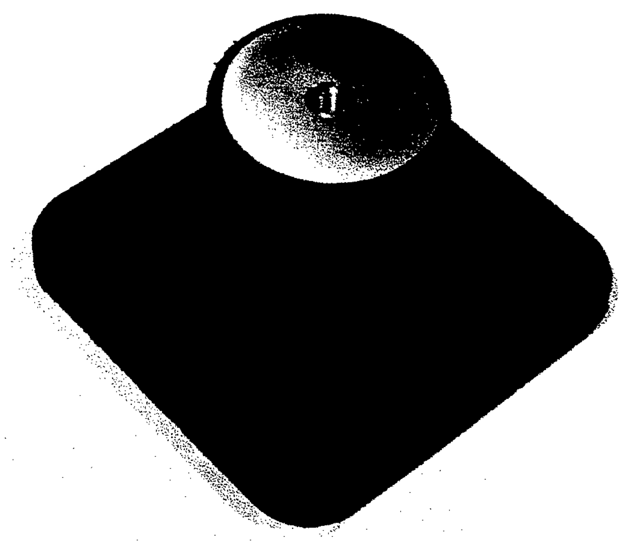

Figure 8 Vibration isolator.

6.1 Test Vibration Isolators

The three test vibration isolators had the same design but employed different rubbers. The three rubbers were natural rubber abbreviated as NR, butyl rubber i.e. poly(isobutene-co-isoprene) abbreviated as IIR, and a low temperature compound abbreviated as LT. The types were Barry Controls GB530-NR2, GB530-B5 and GB530-LT1, which respectively had rubber elements of NR, IIR and LT, and maximum load ratings of 150, 200 and 125 kg.

Prior to testing, each vibration isolator had both ends ground parallel to each other with the minimum removal of material.

6.2 Test Method

The same test method was used for the three vibration isolators, and is described hereunder.

Each vibration isolator was tested with a static load equal to its maximum rating. Figure 6 shows the vibration isolator under test in its normal upright position, with the temperature enclosure removed for clarity. The top of the vibration isolator, i.e. its input, was screwed to an input attachment plate of diameter 170 mm and height 20mm. The bottom of the vibration isolator, i.e. its output, was bolted to an output attachment plate that was square with sides of length 195 mm, and height 20mm. The free ends of the attachment plates were screwed to the force measuring assemblies. The mating surfaces of the vibration isolator, attachment plates and force measuring assemblies were ground flat to ensure intimate contact between them.

The vibration isolator was mechanically conditioned prior to testing by loading and unloading it six times up to the maximum testing static plus dynamic strain plus 30 %. Immediately before testing the vibration isolator was conditioned at a temperature of 13 ±1 °C for 16 hours, and tested at the same temperature.

Because the vibration isolator was asymmetrical, the reversal technique described in Section 4.2 was employed. A logarithm swept sine test was conducted over the frequency range from 5 Hz to 2 kHz with 200 points/decade and 523 points. The vibration isolator was then reversed and a similar test conducted with the same test conditions. Application of equations (6) to (9) to the test data yielded the four-pole parameters. Plots of the measured four-pole parameters are presented in Figures 9 to 20.

The exciting force amplitude was controlled to give a maximum strain of $1\text{x}10^{-3}$ in the rubber element, except over the frequency range from approximately 20 to 80 Hz which had a maximum strain of $3\text{x}10^{-3}$ at approximately 25 Hz.

7. Discussion of Test Results

7.1 Four-Pole Parameters

Consider the four-pole parameters of the vibration isolator with a rubber element of NR, Figures 9 to 12. The vibration isolator comprises a top plate, rubber element and a bottom plate. Let the frequencies of the first troughs of the four-pole parameters α_{11}^*, α_{12}^*, α_{21}^* and α_{22}^* be f_1, f_2, f_3 and f_4 respectively. These frequencies are marked on the Figures.

The first trough of the four-pole parameter α_{11}^* corresponds to the resonance of the top plate on the blocked rubber element, and occurs at a frequency $f_1 = 100$ Hz. Also, the first trough of the four-pole parameter α_{22}^* corresponds to the resonance of the bottom plate on the rubber element blocked at the top, and occurs at a frequency of $f_4 = 66$ Hz. These frequencies depend upon the mass of the appropriate plate, including mass contributions from the rubber element. The mass of the top plate is less than the mass of the bottom plate, including the mass contributions of the rubber element. Therefore the frequency of the first trough of the four-pole parameter α_{11}^* should be greater than that of the four-pole parameter α_{22}^*, which is the case.

The first trough of the four-pole parameter α_{21}^* corresponds to the first standing wave resonance within the rubber element, and occurs at a frequency $f_3 = 253$ Hz. The second standing wave resonance occurs at a frequency of 498 Hz, and has a trough amplitude that is 9 dB lower than that of the first standing wave resonance.

The first trough of the four-pole parameter α_{12}^* corresponds to the resonance of the free top and bottom plates on the rubber element, and occurs at a frequency $f_2 = 124$ Hz. It may be shown that for an ideal vibration isolator,

$$f_2 = \sqrt{f_1^2 + f_4^2} \tag{15}$$

From equation (15) $f_2 = 120$ Hz, which agrees with the measured value of 124 Hz to 3 %. This difference is considered to be experimentally acceptable.

Consider the four-pole parameters of the vibration isolator with a rubber element of IIR, Figures 13 to 16. A similar analysis to that for the NR vibration isolator may be followed. The frequencies of the first troughs of the four-pole parameters α_{11}^*, α_{12}^* and α_{22}^* are approximately $f_1 = 260$ Hz, $f_2 = 310$ Hz and $f_4 = 160$ Hz respectively. The standing waves frequencies are difficult to determine from the figures.

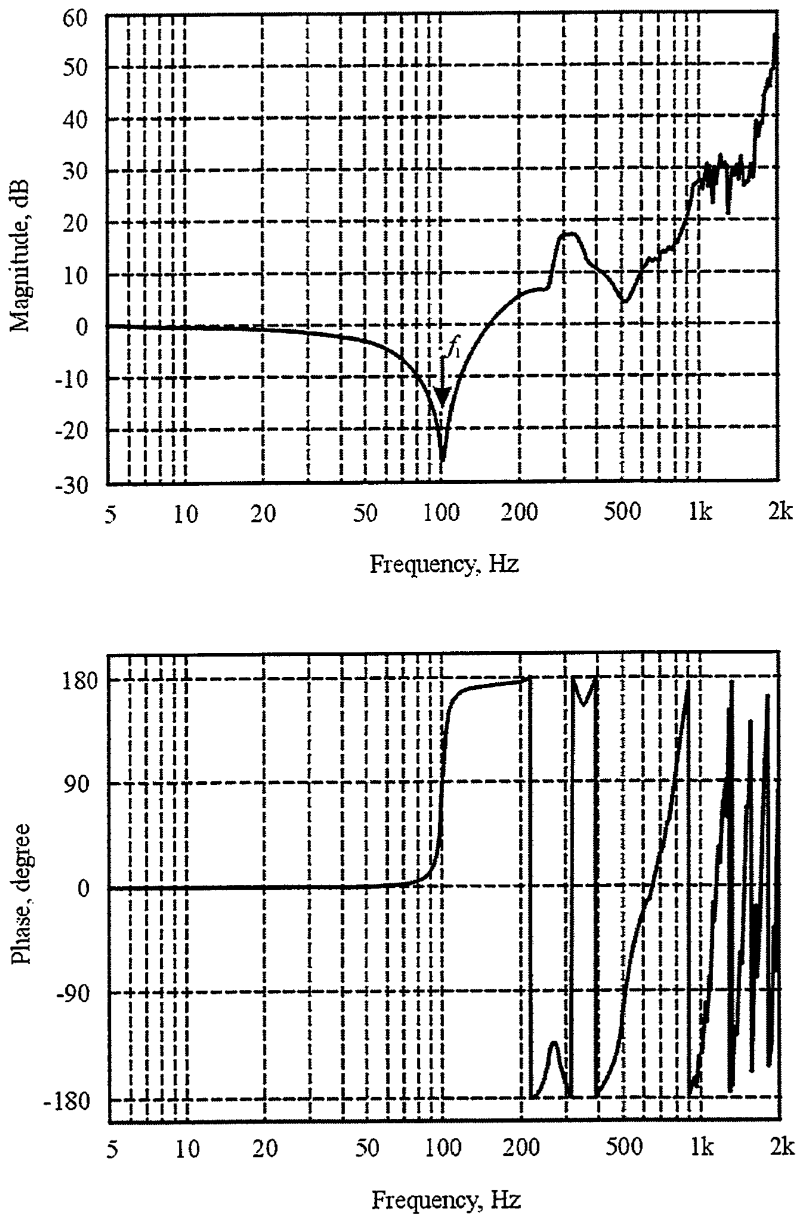

Figure 9 Measured four-pole parameter α_{11}^* for NR vibration isolator.

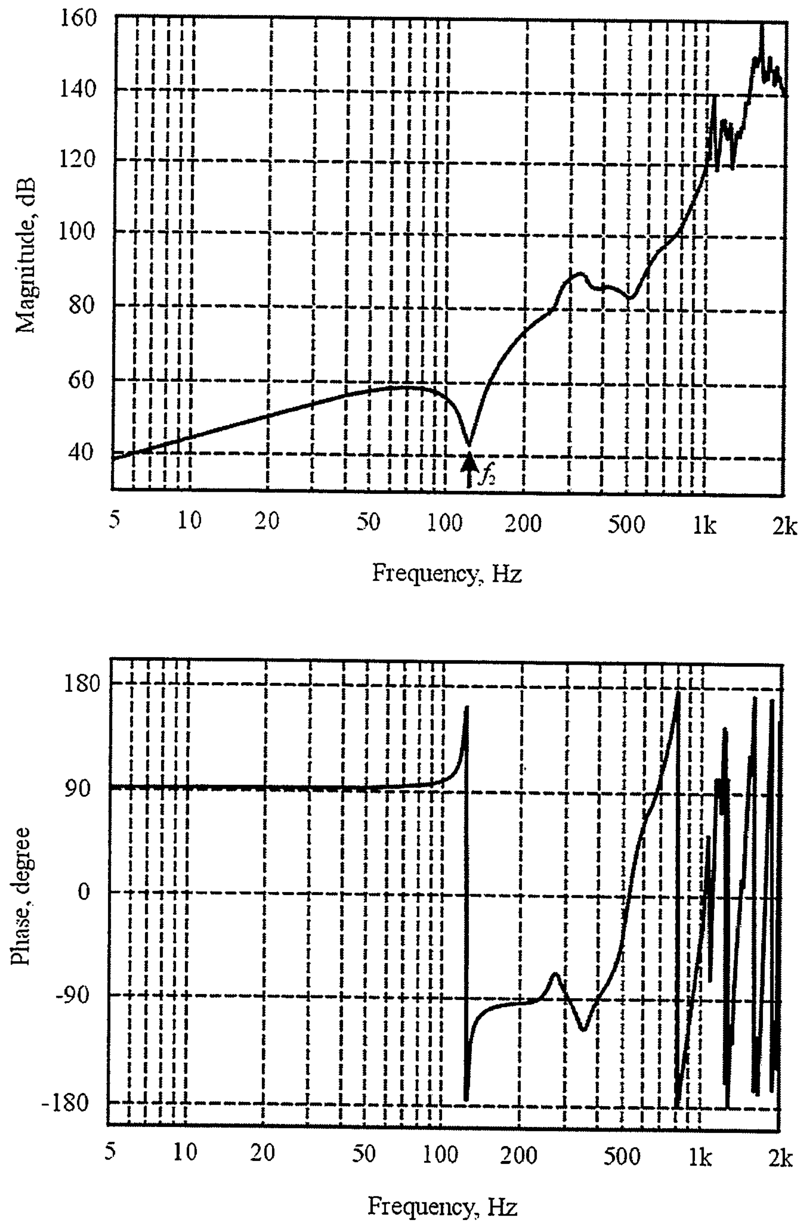

Figure 10 Measured four-pole parameter α_{12}^{*} for NR vibration isolator.

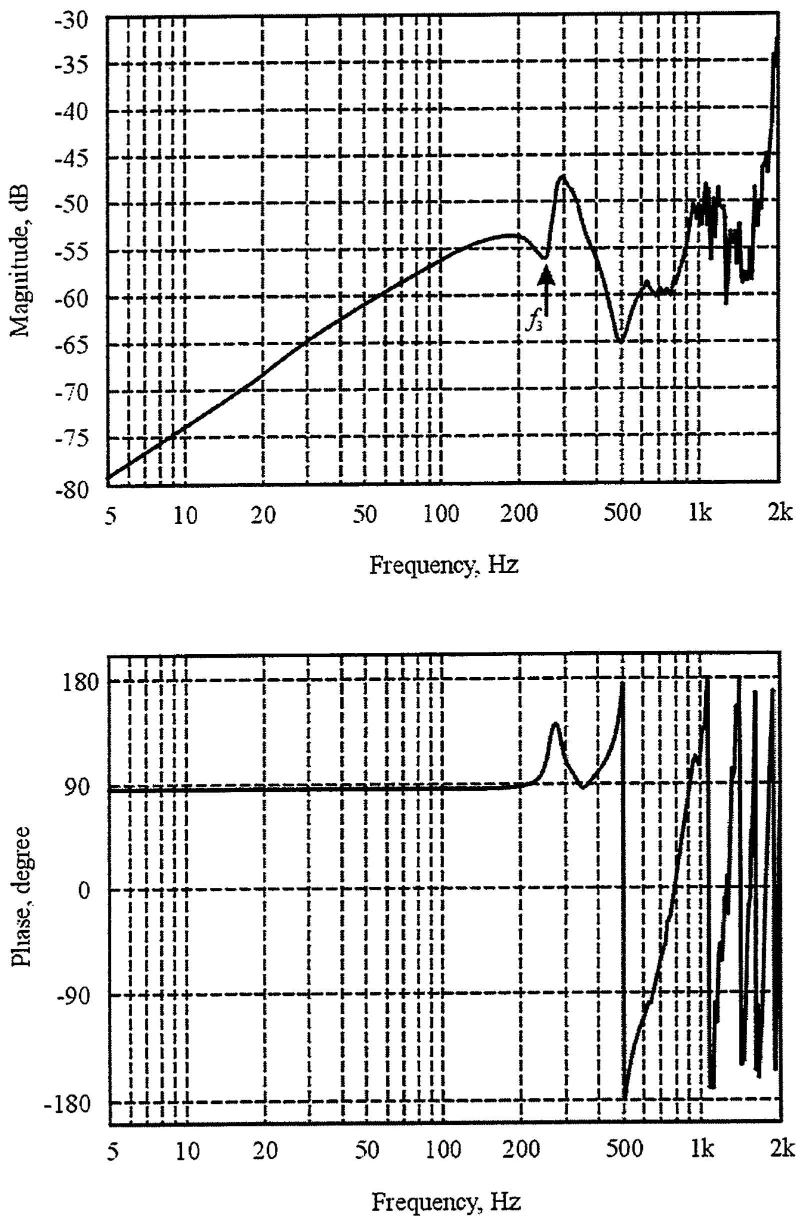

Figure 11 Measured four-pole parameter α_{21}^{*} for NR vibration isolator.

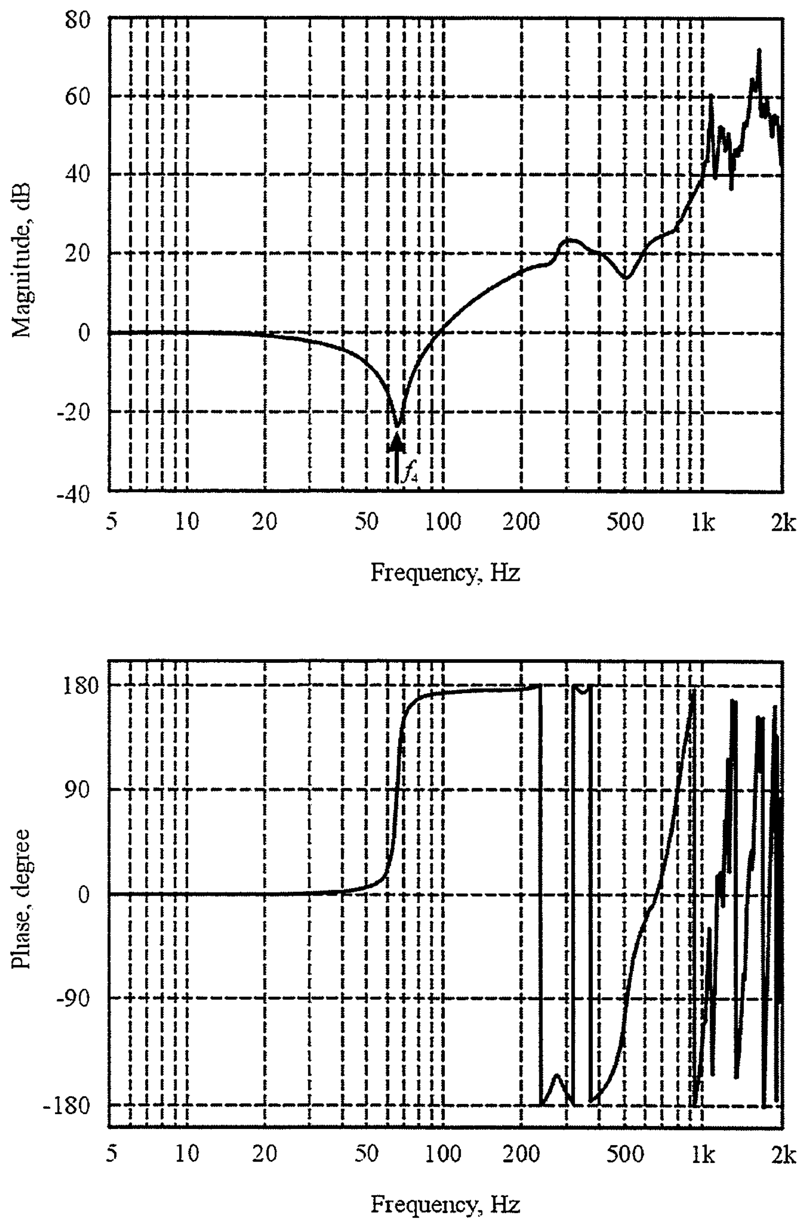

Figure 12 Measured four-pole parameter α_{22}^{*} for NR vibration isolator.

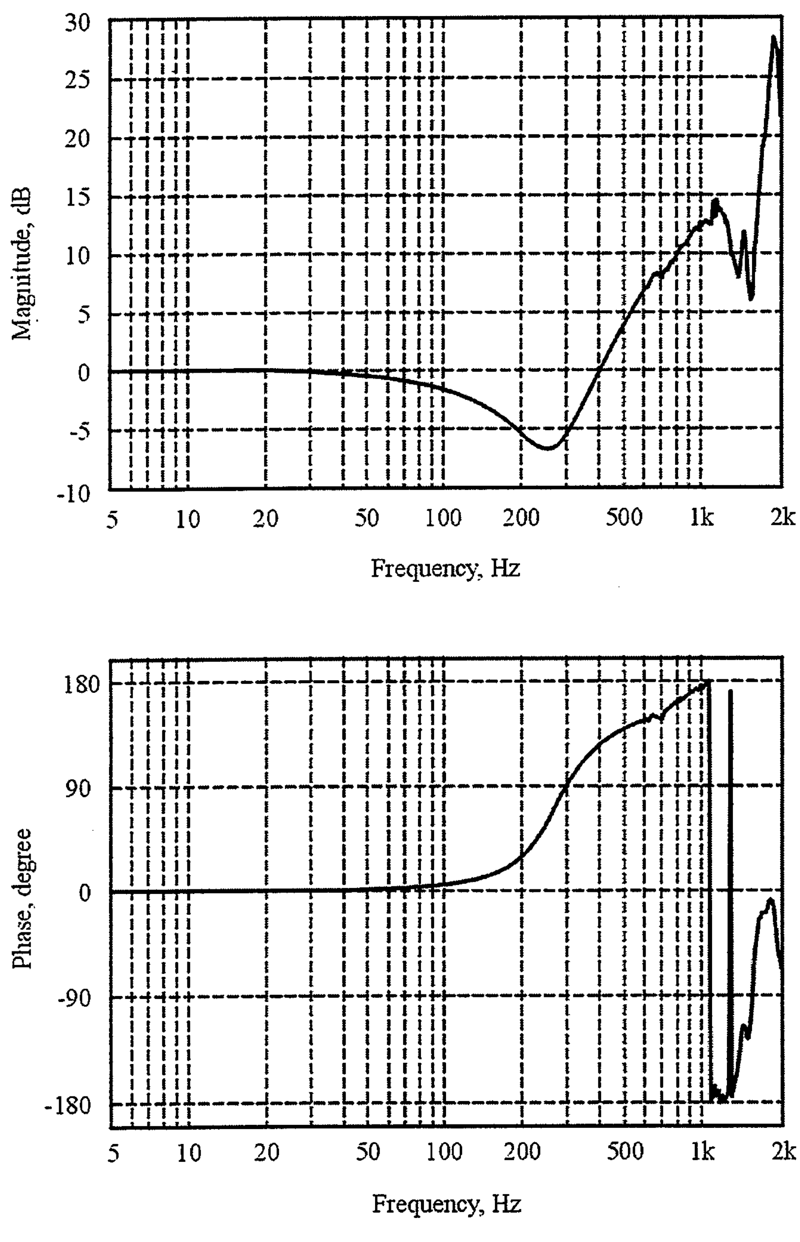

Figure 13 Measured four-pole parameter α_{11}^{*} for IIR vibration isolator.

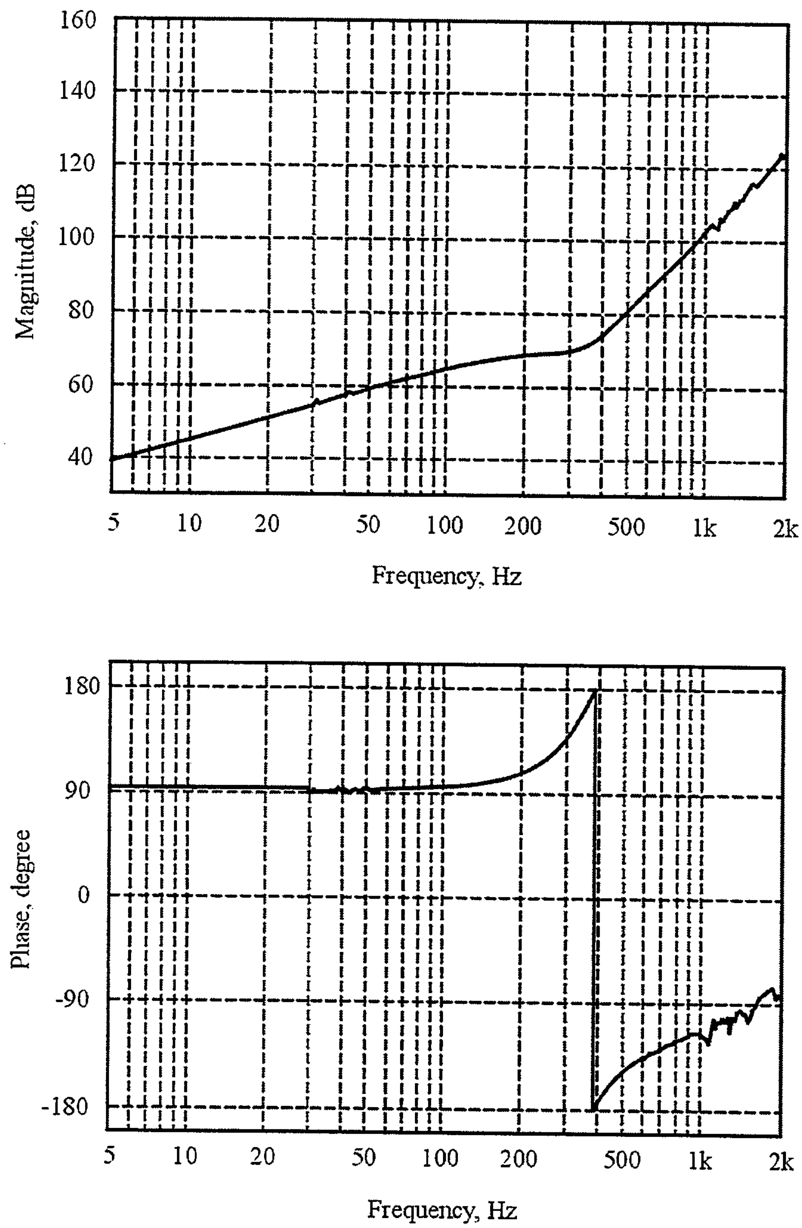

Figure 14 Measured four-pole parameter α_{12}^{*} for IIR vibration isolator.

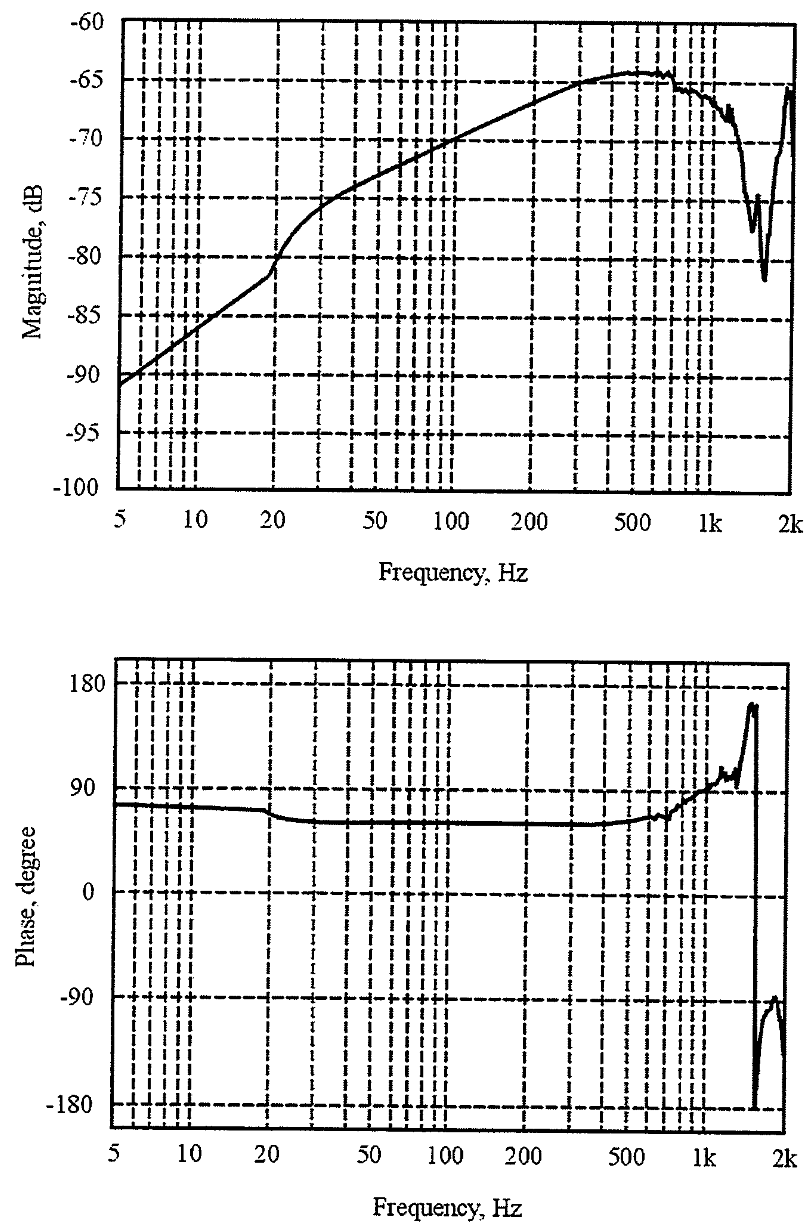

Figure 15 Measured four-pole parameter α_{21}^{*} for IIR vibration isolator.

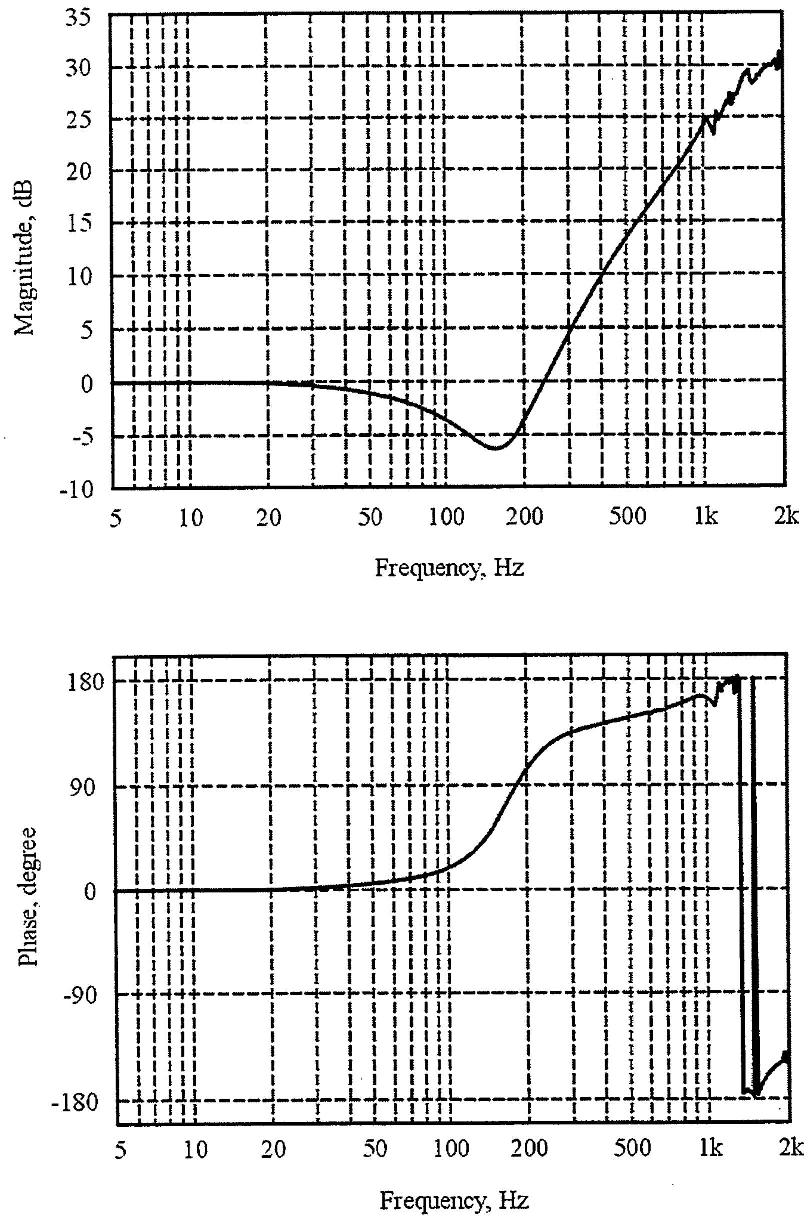

Figure 16 Measured four-pole parameter α_{22}^* for IIR vibration isolator.

Consider the four-pole parameters of the vibration isolator with a rubber element of LT, Figures 17 to 20. A similar analysis to that for the NR vibration isolator may be followed. The frequencies of the first troughs of the four-pole parameters α_{11}^*, α_{12}^*, α_{21}^* and α_{22}^* are $f_1 = 180$, $f_2 = 218$, $f_3 = 486$ and $f_4 = 107$ respectively.

The magnitude of the first troughs of the four-pole parameters α_{11}^* and α_{22}^* is approximately inversely proportional to the loss factor of the vibration isolator. Analysis of these magnitude troughs gives approximate loss factors of 0.06, 0.47 and 0.11 for the NR, IIR and LT rubber elements, respectively. These loss factors were measured at average frequencies of 83, 210 and 144 Hz respectively.

7.2 Effectiveness

The effectiveness of each vibration isolator was calculated for an ideal situation from equation (14), Figure 21. In the ideal situation, each vibration isolator was assumed to support its maximum rated mass, and the foundation was assumed to have zero mobility. Each effectiveness has a minimum that occurs at the resonant frequency f_5 of the supported mass on the vibration isolator. These resonant frequencies are 7, 13 and 13 Hz for the NR, IIR and LT vibration isolators, respectively. In general, the NR vibration isolator has the greatest effectiveness of the three vibration isolators, and is primarily due to its low resonant frequency f_5. Each effectiveness has troughs at the standing wave frequencies.

7.3 Comparison with Manufacturer's Specifications

The manufacturer specifies that the nominal natural frequencies of the vibration isolators are 5 Hz. This means that if the vibration isolator is used to support its maximum rated mass on an idealised foundation of zero mobility, then the mass will have a resonant frequency of 5 Hz. The manufacturer specifies nominal quality-factors (Q-factors) at resonance of 8 to 10, 3.5 and 4 to 5 for the NR, IIR and LT vibration isolators, respectively. These respectively equate to loss factors of approximately 0.12 to 0.1, 0.3 and 0.25 to 0.2.

However, the measured resonant frequencies were 7 Hz, 13 Hz and 13 Hz; and loss factors were 0.06, 0.47 and 0.11; for the NR, IIR and LT vibration isolators respectively. The reasons for the differences between the measured and specified resonant frequencies and loss factors include the effects of vibration amplitude, frequency, temperature and manufacturing variations on the dynamic properties of the rubbers. The effects of vibration amplitude, frequency and temperature are considered in the following sections.

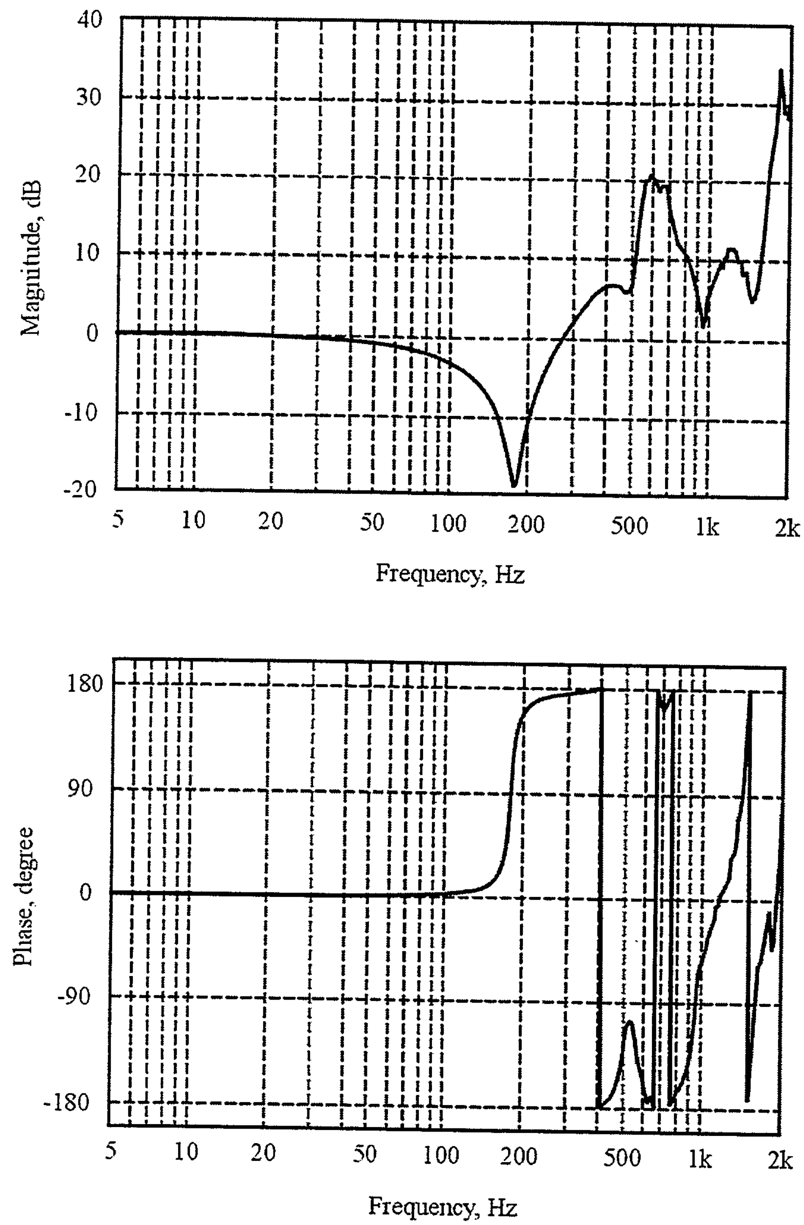

Figure 17 Measured four-pole parameter α_{11}^* for LT vibration isolator.

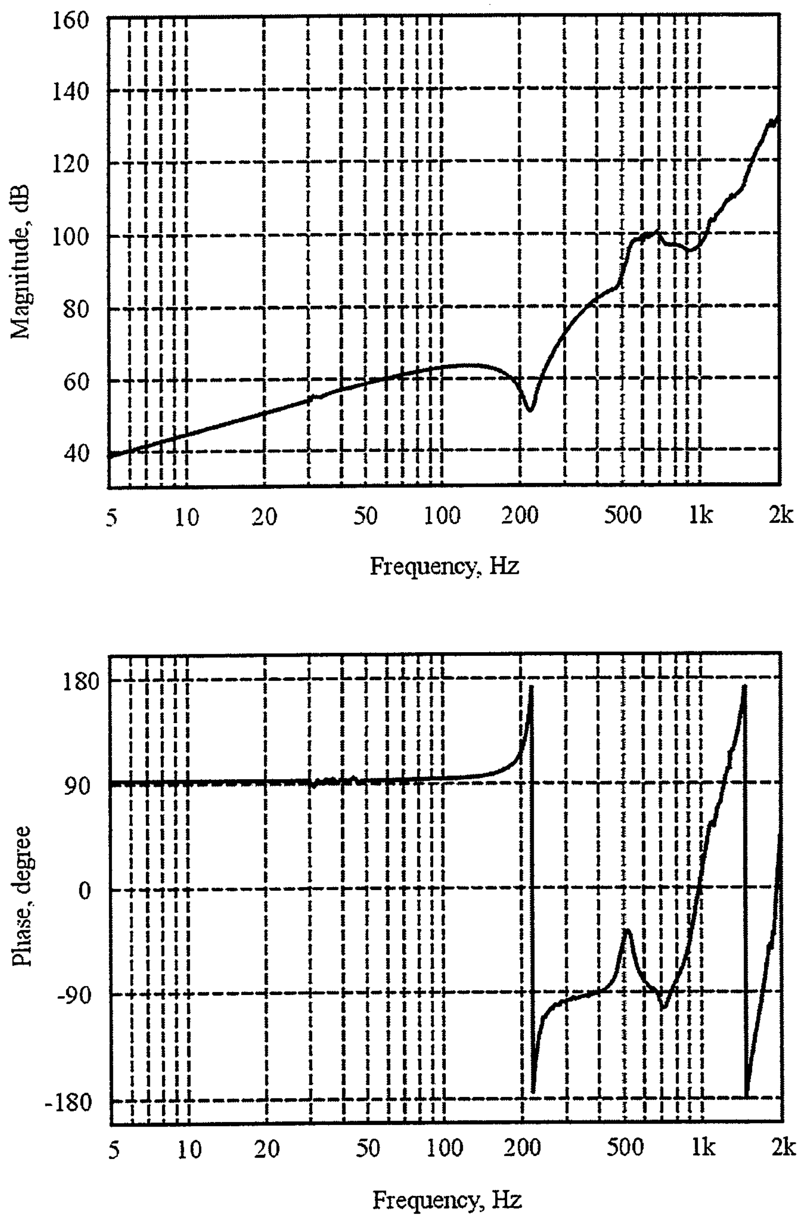

Figure 18 Measured four-pole parameter α_{12}^{*} for LT vibration isolator.

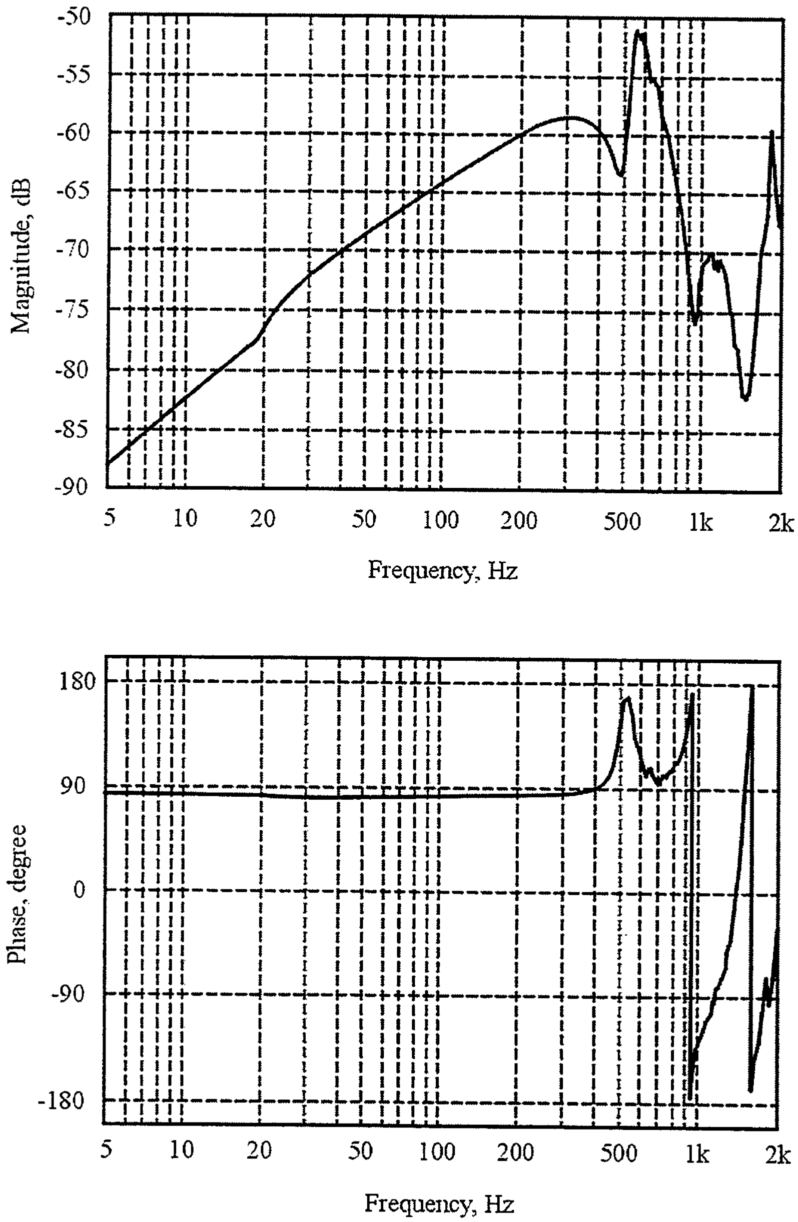

Figure 19 Measured four-pole parameter α_{21}^{*} for LT vibration isolator.

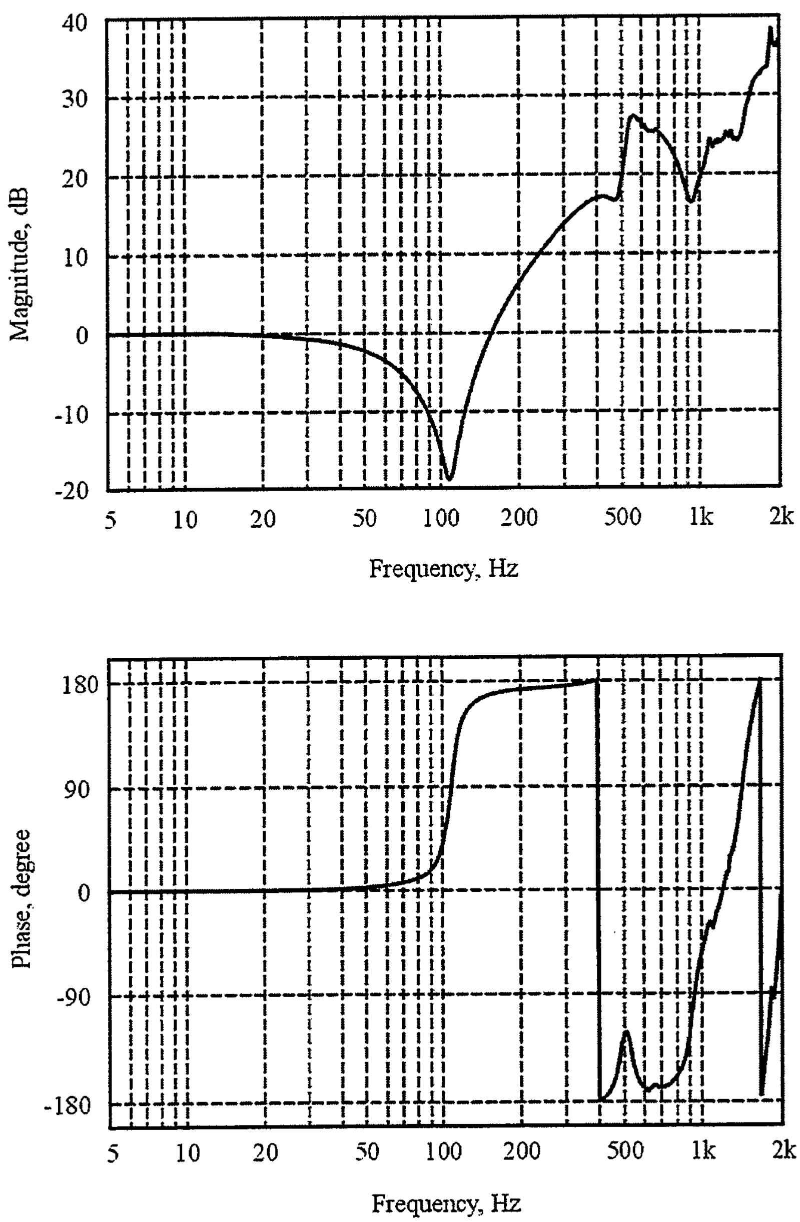

Figure 20 Measured four-pole parameter α_{22}^{*} for LT vibration isolator.

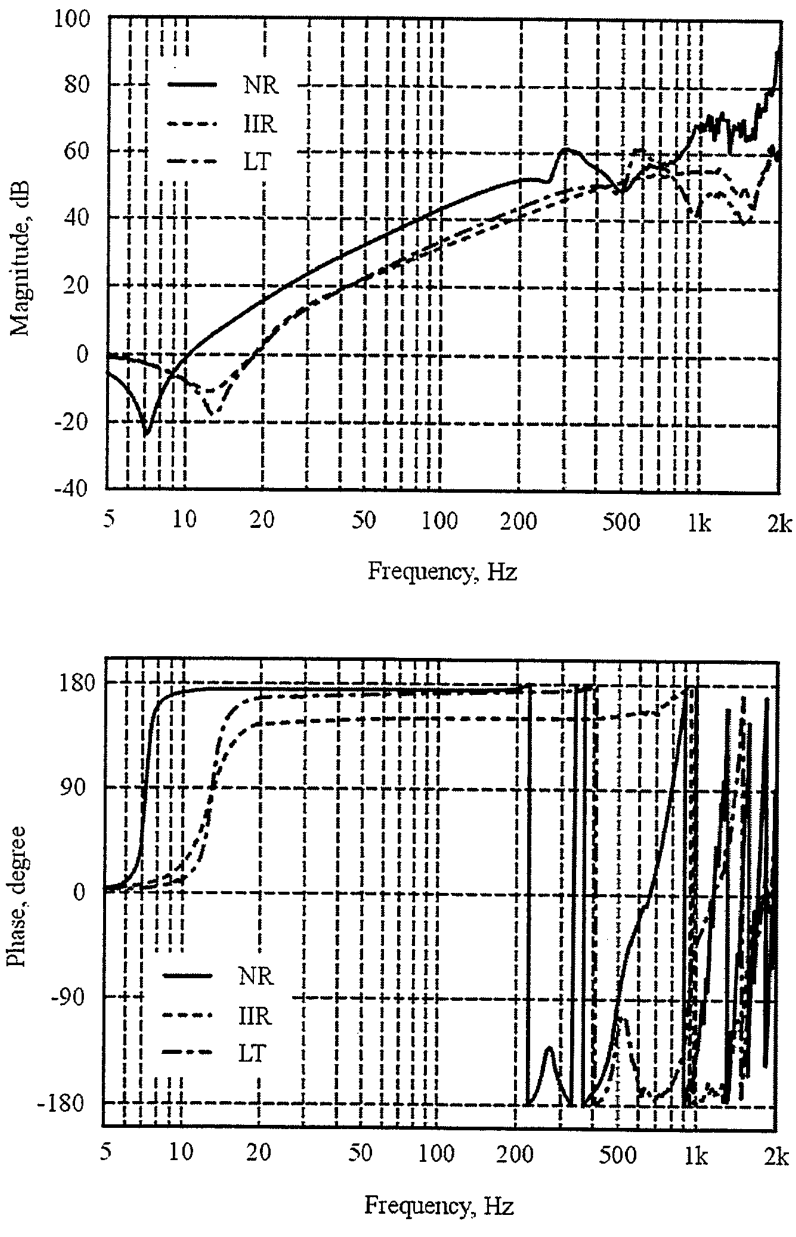

Figure 21 Effectiveness at maximum rated loads.

7.3.1 Effect of Vibration Amplitude

The rubber element of a maritime vibration isolator normally is filled, i.e. incorporates a filler such as carbon black, and is designed to operate in its rubbery plateau region. The complex normal modulus of a filled rubber, operating in its rubbery plateau region, may be treated as constant for small dynamic strains of not greater than $1x10^{-3}$. As the strain increases above $1x10^{-3}$ so the absolute value of the complex normal modulus decreases, Payne (1956) and Payne and Scott (1960). Therefore as the dynamic strain increases above $1x10^{-3}$, so the stiffness magnitude and resonant frequency diminish. It is understood that the manufacturer's specification of the natural frequencies applies for an axial vibration amplitude of 1.0 mm (pk-pk). The effective axial length of the rubber elements of the vibration isolators is approximately 66 mm, and so an axial vibration amplitude of 1.0 mm (pk-pk) approximately equates to a strain of $8x10^{-3}$. Thus the dynamic strain at resonance of approximately $8x10^{-3}$ applied by the manufacturer is larger than the dynamic strain of $1x10^{-3}$ maximum used in the current study. Thus it is expected that the resonant frequencies specified by the manufacturer would be lower than those measured in the current investigation. This is found to be true for all three vibration isolators.

For each vibration isolator, the maximum strain in the rubber element was $1x10^{-3}$ except for the frequency range from approximately 20 to 80 Hz, which had a maximum strain of $3x10^{-3}$ at approximately 25 Hz. The effect of this increased amplitude may be seen in the magnitude of the four-pole parameter α_{21}^*, which displays a small increase above an asymptotic line extended up from the lowest frequencies. Its effect is also visible in the phase of the four-pole parameter α_{21}^*, which undergoes a small decrease below the value at the lowest frequencies. These effects are most evident for the LT vibration isolator, with the magnitude of the four-pole parameter α_{21}^* increasing by approximately 2.4 dB at 25 Hz above the asymptotic line. These effects indicate that the complex normal modulus of the rubber element is dependent on the strain amplitude, and that its magnitude decreases with increasing strain amplitude above $1x10^{-3}$. This agrees with the findings of Payne (1956) and Payne and Scott (1960).

For an idealised vibration isolator, the slope of the asymptotic magnitude of the four-pole parameter α_{21}^* at low frequencies is 20 dB/decade. The measured asymptotic slopes of the tested vibration isolators below 20 Hz was 18 to 19 dB/decade. The difference between the idealised and measured asymptotic slopes is due to the decreasing absolute values of the complex normal moduli of the vibration isolators as the frequency decreases.

Additionally, as the strain increases above $1x10^{-3}$ so the loss factor increases and passes through a maximum, Payne and Scott (1960). Thus it is expected that the loss factors specified by the manufacturer would be higher than those measured in the current investigation. This is found to be true for the NR and LT vibration isolators, but not the IIR vibration isolator.

7.3.2 Effect of Frequency and Temperature

The loss factor increases and passes through a maximum as the frequency increases, for a filled rubber in its rubbery plateau region, Payne and Scott (1960). The manufacturer specified Q-factors at a frequency of 5 Hz, whereas the current study measured the loss factors at frequencies of at frequencies of 83, 210 and 144 Hz for the NR, IIR and LT vibration isolators respectively. However, over the frequency range from 5 to 210 Hz only a small increase in loss factor would be expected, Payne and Scott (1960).

The complex normal modulus of a filled rubber depends upon the temperature, Payne and Scott (1960). For a filled rubber operating in its rubbery plateau region, an increase in temperature causes a reduction in the loss factor. The measurements of this study were conducted at a temperature of 13 ±1 °C. The operating range of temperatures for the vibration isolators was specified from –30 °C to 70 °C. However, the measurement temperatures were not specified by the manufacturer, and so the effect of temperature is unknown. If it is assumed that a "room temperature" of 23 °C was used, then the measured loss factors would be expected to be higher than the specified ones. This effect is particularly noticeable in IIR, Snowdon (1968). This could possibly explain why the measured loss factor was larger than the specified value for the IIR vibration isolator.

7.4 Comparison with Further Testing

Because of the differences observed between the measured and specified dynamic properties, it was decided to conduct a further test for comparison, by employing a different method. As an independent check, the LT vibration isolator was used to support a 100 kg mass above a rigid foundation. As was previously done, the vibration isolator was conditioned at a temperature of 13 ±1 °C for 16 hours, and tested at the same temperature.

The mass was centrally struck using an instrumented modal hammer, and the vertical response of the mass monitored using an accelerometer mounted close to the central axis of the mass. This gave the point inertance, from which the rigid body resonant frequency was determined to be 15.0 Hz. Assuming that the resonant frequency is inversely proportional to the square root of the supported mass, gives a resonant frequency of 13.4 Hz for a 125 kg mass. From the effectiveness calculation of the previous paragraph, the resonant frequency is 13.0 Hz. These frequencies agree to 3 % of each other.

Using the –3 dB bandwidth method on the measured point inertance gave a loss factor of 0.12. From the first troughs of the four-pole parameters α_{11}^* and α_{22}^* the previously measured loss factor was 0.11. Loss factors are difficult to measure accurately, and these measured values have acceptable agreement with each other within experimental error.

Therefore, there is a high degree of confidence in the measured resonant frequencies and loss factors using the vibration isolator test facility.

8. Summary and Conclusions

The dynamic properties of vibration isolators are dependent upon the parameters of static load, amplitude of vibration, frequency and temperature. The study of the three commercial vibration isolators in this paper demonstrated the dependence of their properties and effectiveness on these parameters.

The manufacturer specifies nominal resonant frequencies of 5 Hz, and loss factors of 0.12 to 0.1, 0.3 and 0.25 to 0.2; for the NR, IIR and LT vibration isolators respectively. The measured resonant frequencies were 7 Hz, 13 Hz and 13 Hz; and loss factors were 0.06, 0.47 and 0.11; for the NR, IIR and LT vibration isolators respectively. Thus the study showed that the measured properties differed from the manufacturer's specification. This may be explained because the testing conditions were not the same for the two sets of data. The reasons for the differences include the effects of different vibration amplitudes, frequencies, temperatures and manufacturing variations on the dynamic properties of the rubbers.

It is therefore essential that vibration isolators be tested under service conditions. The characterisations of the vibration isolators using the vibration isolator test facility agreed well with results using an independent method, which gives a high degree of confidence in the measurements using the facility.

The dynamic properties of vibration isolators commonly used in industrial and maritime applications may be measured under service conditions, by employing the vibration isolator test facility. The characterisation of the vibration isolators is in terms of their four-pole parameters, expressed as narrow band amplitude and phase data. The vibration isolator test facility is capable of measuring the four-pole parameters over the frequency range from 5 Hz to 2 kHz, static load range from 1 to 30 kN, and temperature range from 6 to 60 °C. It can test large practical vibration isolators having dynamic stiffness magnitudes of at least 1×10^5 N/m, masses of at least 0.5 kg, and loss factors of at least 0.03.

From the measured four-pole parameters of a vibration isolator, the dynamic properties of any system incorporating it may be investigated. Thus, for example, the effectiveness, force transmissibility and wave effects of a system with a flexible machine and foundation may be determined.

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of Maritime Vibration Isolators

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The development of a novel vibration isolator test facility is described. It is used to measure the dynamic properties of vibration isolators commonly used in maritime and industrial machinery applications. The vibration isolator test facility is capable of measuring the four-pole parameters over the frequency range from 5 Hz to 2 kHz, static load range from 1 to 30 kN, and temperature range from 6 to 60 C.

In order to demonstrate the usefulness of the facility, experimental data for three commercial asymmetrical vibration isolators used for maritime applications are presented. Their four-pole parameters and effectiveness are compared.

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